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Study of a low-temperature Stirling engine driven by a rhombic drive mechanism

Hind El Hassani^{1*}, Nour-eddine Boutammachte², Juergen Knorr³ and Mohamed El Hannaoui¹

Abstract

In this paper, a parametric study of the rhombic drive mechanism of a solar low-temperature Stirling engine was conducted. The goal is to find out the relationship between the rhombic drive mechanism parameters and the performance of a low-temperature differential Stirling engine. The results indicate that the offset distance from the gear center and gear radius of the rhombic drive mechanism have to be maximized, in order to increase the indicated power, while other rhombic drive parameters should be reduced for the same reason. Another result is that for realizing the optimized phase angle for beta-type Stirling engines, one should act on working piston-related bars and rhombic drive gear radius.

Keywords: Rhombic drive; Stirling engine; Indicated power; Phase angle

Background

The Stirling engine is an externally heated machine which runs according to a reversible closed cycle. It was invented in 1816 by the Stirling brothers [1]. This engine is known by high heat conversion efficiency, reliability, low noise operation, and ability to use many fuels. These characteristics make the Stirling engines adapted to the demand of the effective use of energy and environmental preservation [2,3]. In fact, reducing environmental impacts of conventional energy resources and meeting the growing energy demand of the global population had motivated considerable research attention in a wide range of environmental and engineering application of renewable form of energy [4], and among all possible alternative energy options, solar energy is becoming more popular in the world. This is mainly due to the availability of plenty of sunlight in many countries [5].

The low-temperature differential Stirling engine (LTD-SE) is a kind of Stirling engine that can run with a small temperature difference between the hot and the cold sources. In 1983, Kolin [6] demonstrated the first LTD-SE. Senft [6,7] developed the Ringbom engine using Kolin's conclusions. Many other prototypes of LTD-SE

were manufactured and tested. In 2006, Kongtragool and Wongwises [8] constructed twin power pistons and four power pistons, gamma-configuration LTD Stirling engines. In 2007, Martaj et al. [9] presented a thermodynamic analysis of a low-temperature Stirling engine at steady state operation. In 2012, Boutammachte and Knorr [10] studied a γ -type LTD-SE. They presented several experimental measurements made under laboratory and field-test conditions in Morocco [11]. Measurements with flat plate cooler and discontinuous motion of the displacer were conducted to verify some recommendations of Kolin in expectation of power output improvement. They concluded that the assumption of Kolin related to the proportionality of speed to temperature difference is justified, but the expectation about improvement of the machine performance with discontinuous motion of the two pistons needs to be reviewed.

As a result of a scientific cooperation between Moulay Ismail University in Morocco and the University of Technology of Dresden in Germany, an LTD γ -type Stirling engine was manufactured. This engine was tested and studied under real conditions [11]. Based on the conclusions of this study, a new prototype of LTD-SE with rhombic drive mechanism is being developed. This new prototype is designed to be simple, robust, cheap, and stable to be used for water pumping in poor rural African areas. In fact, this engine is manufactured more

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than 90% from mechanical cheap parts available in poor areas, so that it could be easily manufactured and maintained right there. Besides, it is a solar engine, fact that lowers significantly its use price. In addition to that, the new prototype is a beta type which has a symmetric geometry that gives the engine more stability compared to γ -type Stirling engines, and the drive mechanism used, which is the rhombic drive, allows the coaxial position of the engine pistons, fact that improves also the engine stability.

Since this machine is developed to be as low expensive as possible, it was designed without any concentrator mechanism; consequently, its power output is relatively low. Therefore, a special attention should be kept for each construction parameter in order to optimize the machine performance.

In this optic, this paper aims to study the drive mechanism adopted in the studied machine, which is the rhombic drive mechanism. Thus, a parametric study based on Schmidt theory is conducted to find out the relationship between the rhombic drive parameters and the performance of low-temperature Stirling engines. A thermodynamic analysis of the prototype is also conducted. Moreover, as the relationship between rhombic drive mechanism parameters is not enough studied and available for researchers, a study of these parameters is conducted, and a methodology of choosing their values is suggested.

Methods

Prototype description

As a result of a scientific cooperation between Dresden University of Technology and Moulay Ismail University in Morocco, an LTD-SE (beta type) is under construction.

The mechanical arrangement of the manufactured LTD-SE and its specifications are shown in Figure 1 and Table 1.

Kinematic relations

Figure 2 shows the parameters of the rhombic drive mechanism. The position and velocity vectors of each link are expressed as follows:

$$\vec{oa} = (r_g + e \cos \phi) \vec{i} + e \sin \phi \vec{j} \quad (1)$$

$$\vec{v}_a = \frac{d \vec{oa}}{dt} = -e \omega \sin \phi \vec{i} + e \omega \cos \phi \vec{j} \quad (2)$$

$$\vec{ob} = (r_g + e \cos \phi + d_1 \cos \phi_1) \vec{i} + (e \sin \phi + d_1 \sin \phi_1) \vec{j} \quad (3)$$

$$\vec{v}_b = \frac{d \vec{ob}}{dt} = (-e \omega \sin \phi - d_1 \omega_1 \sin \phi_1) \vec{i} + (e \omega \cos \phi + d_1 \omega_1 \cos \phi_1) \vec{j} \quad (4)$$

$$\vec{ob'} = \vec{oa} + \vec{ab'} = (r_g + e \cos \phi + d_2 \cos \phi_2) \vec{i} + (e \sin \phi + d_2 \sin \phi_2) \vec{j} \quad (5)$$

$$\vec{v}_{b'} = \frac{d \vec{ob'}}{dt} = (-e \omega \sin \phi - d_2 \omega_2 \sin \phi_2) \vec{i} + (e \omega \cos \phi + d_2 \omega_2 \cos \phi_2) \vec{j} \quad (6)$$

From the geometrical constraints,

$$r_g + e \cos \phi + d_1 \cos \phi_1 = d_3/2 \quad (7)$$

$$r_g + e \cos \phi + d_2 \cos \phi_2 = d_4/2 \quad (8)$$

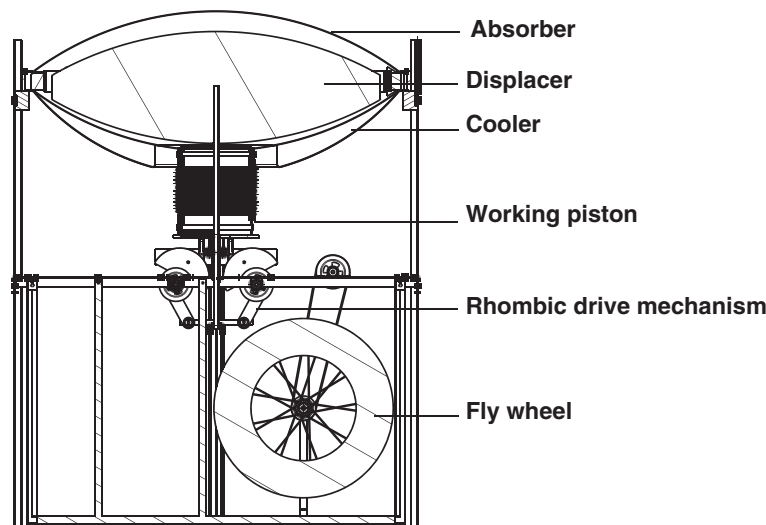


Figure 1 Schematic illustration of the prototype.

Table 1 Technical specifications of the prototype (baseline case)

Parameters	Description/value
Engine type	β
Working fluid	Air
Cooling system	Water cooled
Displacer diameter	1.35 m
Working piston diameter	0.29 m
Absorber diameter	1.51 m
Speed	30 rpm
Absorber temperature	70°C
Cooler temperature	20°C
Regenerator volume	0.063 m ³

$$\phi_1 = \cos^{-1} \left[\frac{1}{d_1} \left(\frac{d_3}{2} - r_g - e \cos \phi \right) \right] \quad (9)$$

$$\phi_2 = \cos^{-1} \left[\frac{1}{d_2} \left(\frac{d_4}{2} - r_g - e \cos \phi \right) \right] \quad (10)$$

Then, the y position of the working piston and the displacer are as follows:

$$Y(\text{working piston}) = L_d + e \sin \phi + d_1 \sin \phi_1 \quad (11)$$

$$Y(\text{displacer}) = L_p + e \sin \phi + d_2 \sin \phi_2 \quad (12)$$

Based on the prototype data (baseline case), available in Tables 1 and 2, one can get the vertical positions of the working piston and displacer (Figure 3) and variation of engine spaces versus crank angle (Figure 4).

Engine work and power

Schmidt [12] presented a mathematically exact expression for calculating the indicated work of Stirling engines. The Schmidt theory adopts many ideal assumptions, like perfect regeneration, isothermal compression and expansion, and harmonic motion of the displacer and the working piston. The Schmidt theory is widely used due to its mathematical simplicity, and it is considered by many researchers, suitable for studying Stirling engines in designing stage, as it is the case for this study.

Schmidt first calculated the gas mass in different parts of the machine and used the perfect gas equation. Applied on our prototype (Figure 5, Table 3), we obtain the following equations:

$$p = mr \left(\frac{V_c}{T_c} + \frac{V_f}{T_f} + \frac{V_{Tr}}{T_f} + \frac{V_r}{T_r} + \frac{V_{mc}}{T_c} + \frac{V_{mf}}{T_f} + \frac{V_{mTr}}{T_f} \right)^{-1} \quad (13)$$

Schmidt adopted the assumption of harmonic motion:

$$V_c = \frac{V_{sc}}{2} (1 + \cos(\theta + \alpha)) \quad (14)$$

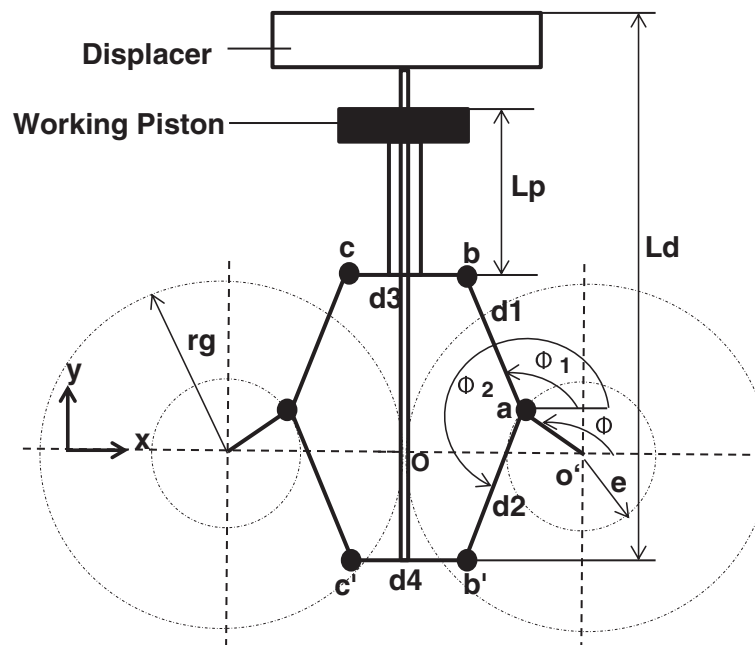


Figure 2 Schematic illustration of the rhombic drive mechanism parameters.

Table 2 Geometrical variables of the rhombic drive mechanism (baseline case)

	Value (m)
e	0.035
d_1	0.125
d_2	0.127
d_3	0.228
d_4	0.226
rg	0.18
Ld	0.808
Lp	0.112

$$V_f = \frac{V_{SC}}{2} (1 - \cos(\theta + \alpha)) \quad (15)$$

$$V_{TR} = \frac{V_{STR}}{2} (1 + \cos\theta). \quad (16)$$

Table 3 Nomenclature and indices

	Description
Nomenclature	
C_p	Specific heat at $p = \text{cst}$ (J/kg K)
C_v	Specific heat at $v = \text{cst}$ (J/kg K)
e	Offset distance from the gear center (m)
h	Specific enthalpy (J/kg)
Ld	Displacer length (m)
Lp	Working piston length (m)
n	Engine speed (Hz)
P	Pressure (Pa)
Q	Heat (J)
r	Specific gas constant (J/kg K)
rg	Rhombic drive gear radius (m)
T	Temperature (K)
V	Volume (m ³)
V_{SC}	Maximal volume of hot space (m ³)
V_{STR}	Maximal volume of working space (m ³)
W	Work (J)
α	Phase angle
Φ	Crank angle
η	Efficiency
Indices	
c	Hot space
f	Cold space
m	Dead (volume)
r	Regenerator
th	Thermal
TR	Working space

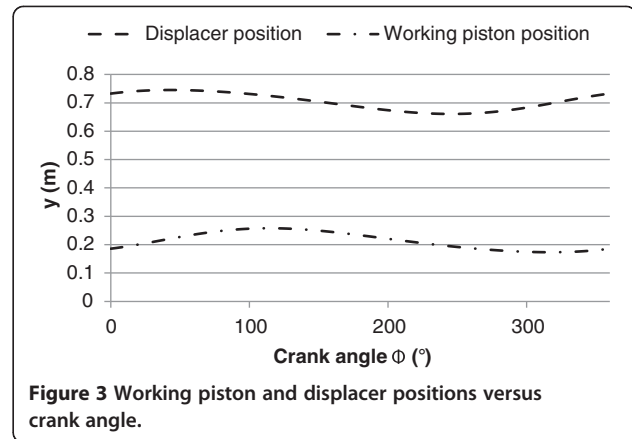


Figure 3 Working piston and displacer positions versus crank angle.

The pressure is expressed as follows:

$$P = \frac{mr/k}{1 + \left(\frac{A}{k}\right) \cos(\theta - \beta)} \quad (17)$$

$$k = \frac{V_{SC}}{2Tc} + \frac{V_{SC}}{2Tf} + \frac{V_{STR}}{2Tf} + \frac{2V_r}{Tf + Tc} + \frac{V_{mc}}{Tc} + \frac{V_{mf}}{Tf} + \frac{V_{mTR}}{Tf} \quad (18)$$

$$A = \frac{1}{2} \left[\left(\frac{V_{STR}}{Tf} \right)^2 + \left(\frac{V_{SC}}{Tc} - \frac{V_{SC}}{Tf} \right)^2 - 2 \frac{V_{STR}}{Tf} \left(\frac{1}{Tf} - \frac{1}{Tc} \right) V_{SC} \cos\alpha \right]^{\frac{1}{2}} \quad (19)$$

$$\tan\beta = \frac{\left(\frac{1}{Tf} - \frac{1}{Tc} \right) \sin\alpha}{\frac{V_{STR}}{Tf \times V_{SC}} - \left(\frac{1}{Tf} - \frac{1}{Tc} \right) \cos\alpha} \quad (20)$$

Cyclic work and power are expressed as follows:

$$W_{\text{tot}} = W_{TR} = \int_0^{2\pi} P dV_{TR} = -\frac{1}{2} V_{STR} \int_0^{2\pi} P \sin\theta d\theta \quad (21)$$

$$W_{\text{tot}} = V_{STR} \frac{\pi m R}{A} \left(\frac{1}{\sqrt{1-k_1^2}} - 1 \right) \sin\beta \quad (22)$$

$$P = \frac{n \times W_{\text{tot}}}{2\pi} = V_{STR} \frac{n m R}{2A} \left(\frac{1}{\sqrt{1-k_1^2}} - 1 \right) \sin\beta, \quad (23)$$

where n is the engine speed and m is the total gas mass.

Thermodynamic analysis

Applying the thermodynamic analysis proposed by Martaj et al. [9] to the particular studied prototype, we obtain the following results.

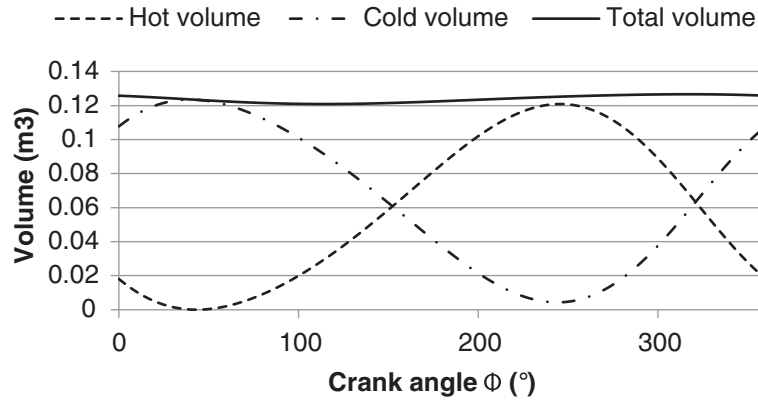


Figure 4 Variation of engine spaces versus crank angle.

Regenerator analysis

The regenerator is used to store and release the heat exchanged with the fluid during its movement from the cold cell towards the hot cell and conversely. Schmidt assumes that the regeneration is perfect. But in practice, it is difficult to realize a perfect regenerator. In this thermodynamic analysis, it is supposed an imperfect regeneration. Thus, the fluid temperature at the exit of the regenerator towards the cold cell $T_{f'}$ is higher than T_f and the fluid temperature at the exit of the regenerator towards the hot cell $T_{c'}$ is lower than T_c (Figure 6).

The regenerator efficiency is defined by

$$\eta_r = \frac{T_c - T_{f'}}{T_c - T_f} = \frac{T_{c'} - T_f}{T_c - T_f} = 1 - \frac{\Delta Tr}{T_c - T_f}, \quad (24)$$

where ΔTr represents the temperature pinch in the regenerator, assumed identical at the two extreme orifices of the regenerator:

$$\Delta Tr = T_{f'} - T_f = T_c - T_{c'}. \quad (25)$$

Since the regenerator volume is constant, the work exchanged is null, and the average temperature Tr is supposed to be constant.

$$Tr = \frac{T_{f'} + T_c}{2}. \quad (26)$$

The first thermodynamic principle for an open system is written as

$$\delta Q + \delta W + \sum h_i dm_i = C_v d(mT). \quad (27)$$

In the regenerator cell, Equation 27 can be written as

$$\delta Q_r = C_p T_{f'} dm_f + C_p T_{c'} dm_c + \frac{C_v}{r} V_r dp, \quad (28)$$

where dm_i is positive when it enters the volume i . Two cases arise for each opening:

- Flow from cold side towards the regenerator: $dm_f < 0$, then $T_{f'} = T_f$, $T_{c'} = T_c$.
- Flow from hot side towards the regenerator: $dm_c < 0$, then $T_{c'} = T_c$, $T_{f'} = T_f$.

The mass in this cell is

$$m_r = \frac{PV_r}{rT_r}. \quad (29)$$

The mass variation in the regenerator is

$$dm_r = -dm_c - dm_h = \frac{V_r}{rT_r} dp. \quad (30)$$

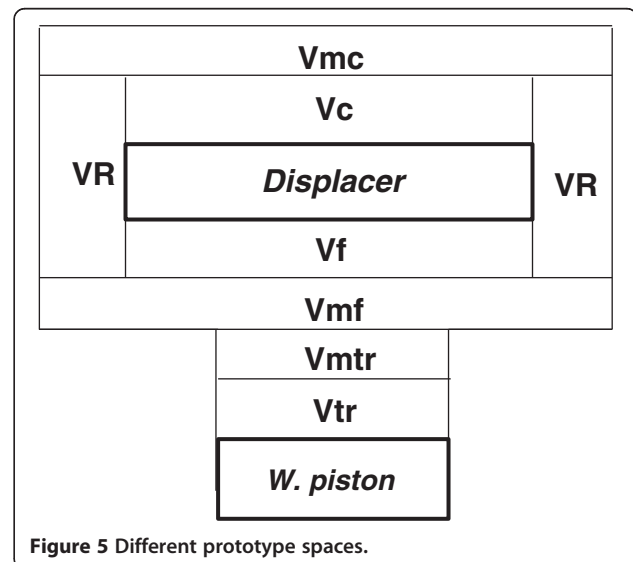


Figure 5 Different prototype spaces.

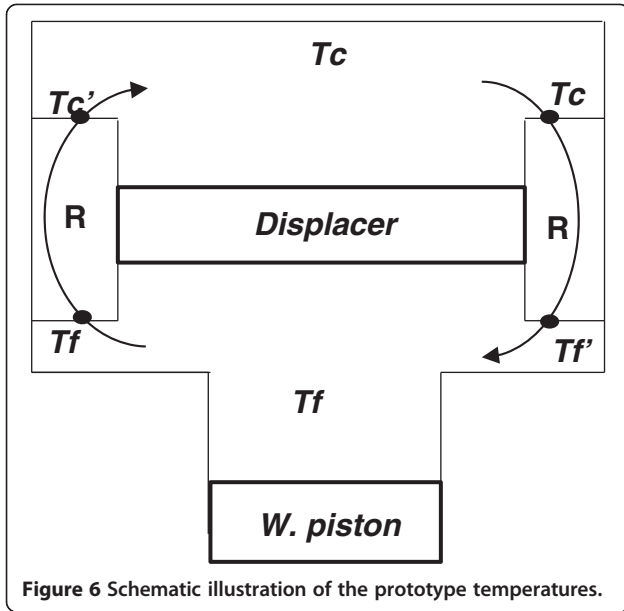


Figure 6 Schematic illustration of the prototype temperatures.

Heating cell

The heating cell has a single communication orifice; Equation 27 could be written in the following form:

$$\delta Q_c + \delta W_c + C_p T_{rc} dm_c = C_v \cdot d(mT)_c. \quad (31)$$

As stated before, if elementary mass is entering the hot volume ($dm_c > 0$), then $T_{rc} = T_{c'}$, else $T_{rc} = T_c$.

The differential form of the perfect gas state equation is

$$\frac{dp}{p} + \frac{dV_c}{V_c} = \frac{dm_c}{m_c} + \frac{dT_c}{T_c}. \quad (32)$$

The hot cell is supposed to be isothermal; thus, Equation 33 becomes

$$\frac{dp}{p} + \frac{dV_c}{V_c} = \frac{dm_c}{m_c}. \quad (33)$$

According to the perfect gas state equation,

$$m_c = \frac{PV_c}{rT_c}$$

and

$$dm_c = \frac{V_c dp}{T_c r} + \frac{P dV_c}{T_c r} = \frac{1}{rT_c} (V_c dp + P dV_c). \quad (34)$$

Equation 31 becomes

$$\delta Q_c = -C_p T_{rc} dm_c + \frac{C_v}{r} V_c dp + \frac{C_p}{r} P dV_c \quad (35)$$

$$\delta Q_c = V_c \frac{C_p}{r} \left(\frac{1}{\gamma} - \frac{T_{rc}}{T_c} \right) dp + P \frac{C_p}{r} \left(1 - \frac{T_{rc}}{T_c} \right) dV_c. \quad (36)$$

Cooling cell

Using the same reasoning as for the previous cell, Equation 27 becomes

$$\delta Q_f = -C_p T_{rf} dm_f + \frac{C_v}{r} V_f dp + \frac{C_p}{r} P dV_f. \quad (37)$$

If $dm_f > 0$, then $T_{rf} = T_f$, else $T_{rf} = T_f$

As for the heating cell, we obtain also

$$dm_f = \frac{V_f dp}{T_f r} + \frac{P dV_f}{T_f r} = \frac{1}{rT_f} (V_f dp + P dV_f). \quad (38)$$

Thus, Equation 37 becomes

$$\delta Q_f = V_f \frac{C_p}{r} \left(\frac{1}{\gamma} - \frac{T_{rf}}{T_f} \right) dp + P \frac{C_p}{r} \left(1 - \frac{T_{rf}}{T_f} \right) dV_f. \quad (39)$$

Engine balance

During a crankshaft rotation, the heat exchanged in the cooling and heating cells are $Q_c = \oint \delta Q_c$ and $Q_f = \oint \delta Q_f$ obtained by integration of Equations 36 and 39. Since the regenerator is supposed imperfect, an additional quantity of heat Q_r to bring by the hot source, is essential:

$$\oint \delta Q_r = \oint C_p T_{rf} dm_f + \oint C_p T_{rc} dm_c. \quad (40)$$

The overall engine balance is written as

$$\begin{aligned} W + Q_f + Q_c + Q_r &= 0 \\ W &= -(Q_f + Q_c + Q_r). \end{aligned} \quad (41)$$

The work is carried out by the working piston during compression and expansion. It can also be calculated by the following expression:

$$W = W_f + W_c. \quad (42)$$

This work is also

$$W = -\oint P dV, \quad (43)$$

where dV is the total fluid variation and P is the internal pressure. The thermal engine efficiency is expressed by

$$\eta_{th} = \frac{W}{Q_c + Q_r}. \quad (45)$$

Rhombic drive study

The rhombic drive mechanism was proposed by the Philips Company [13] and widely used for beta-type Stirling engines [14]. The major advantage of the rhombic drive mechanism is that the coaxial movement of the piston and the displacer is quiet and requires no serious lubrication [15].

As the relationship between rhombic drive mechanism parameters is not studied enough and available for researchers, a study of these parameters was conducted. Here are some conditions for good functioning of the rhombic drive mechanism:

1. At $\Phi = 0^\circ$, the rigid rods d_1 and d_2 should be tall enough in order to reach the most far point during the cycle:

$$d_1 > rg - \frac{d_3}{2} + e \quad (46)$$

$$d_2 > rg - \frac{d_4}{2} + e. \quad (47)$$

2. At $\Phi = 180^\circ$, gears should be far enough from each other:

$$rg > e + \frac{d_3}{2} \quad (48)$$

$$rg > e + \frac{d_4}{2}. \quad (49)$$

As a consequence, during the design of the rhombic drive mechanism, we propose the following order to choose the rhombic drive parameters:

Choosing the crossbar lengths d_3 and d_4 and the offset distance from the gear center e

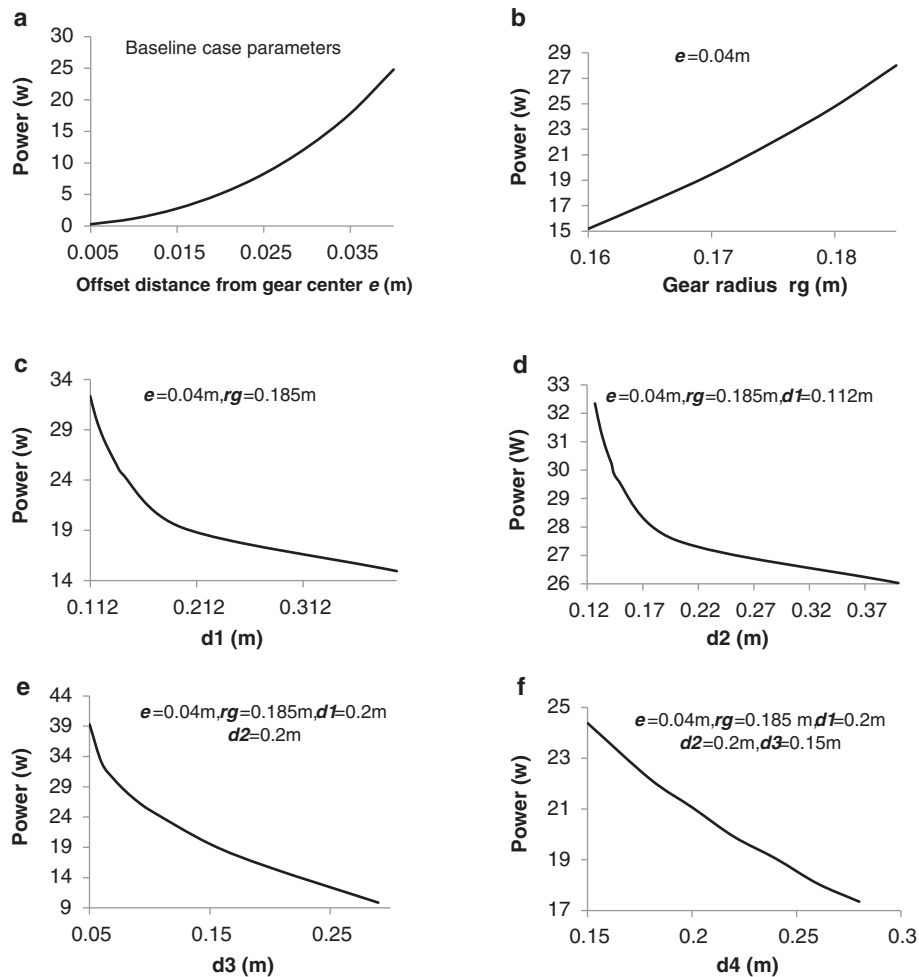


Figure 7 Effect of rhombic drive parameters on power.

1. Calculating the interval of values allowed to gear radius as follows:

$$\min \left(e + \frac{d_3}{2}, e + \frac{d_3}{2} \right) < rg < \max \left(d_2 + \frac{d_4}{2} - e, d_1 + \frac{d_3}{2} - e \right). \quad (50)$$

2. Choosing the connecting bar lengths d_1 and d_2 , with respect to the conditions below:

$$d_1 > rg - \frac{d_3}{2} + e \quad (51)$$

$$d_2 > rg - \frac{d_4}{2} + e. \quad (52)$$

A parametric study based on the theoretical analysis is performed in order to evaluate the effect of each rhombic drive parameter on the machine characteristics. Note

that Schmidt calculating assumptions are adopted. At each study, only one parameter is changed in order to find out its single effect. Results are found using a computer program on the Microsoft EXCEL. For a complete revolution of the crank and at a step of 1° , the program computes the instantaneous positions of the displacer and the working piston, the volume of each space of the machine, and the total mass, work, and power of the cycle. The program has the ability to change power calculation parameters in order to show their single effect.

The geometrical and the operating variables which are regarded as the baseline case in this study are listed in Tables 1 and 2. In this case, the rotational speed is set at 30 rpm; the absorber and the cooler temperatures are set at 70°C and 20°C , respectively.

Results and discussion

Figure 7 displays the effect of rhombic drive parameters on indicated power. Figure 7a,b shows that any increase in offset distance from the gear center (e) or gear radius

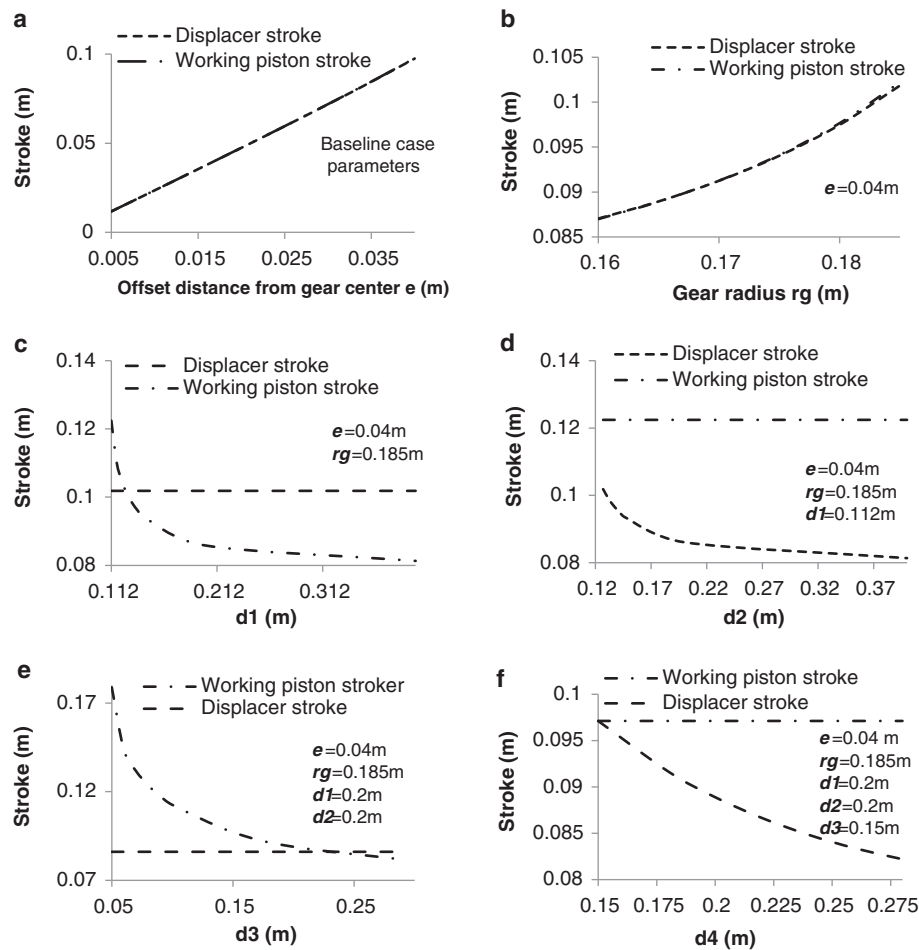


Figure 8 Effect of rhombic drive parameters on piston strokes.

(rg) has a positive influence on power. Figure 8a,b gives the explanation of the former fact; when increasing e and rg, strokes of displacer and also working piston increase, then dead volumes are reduced, and total volume variation is increased. All of these have direct positive influence on indicated power. It is noticed that for the same variation of e and rg ($\Delta e \approx \Delta rg \approx 0.025$ m), the offset distance from gear e has larger positive effect on the indicated power than rg ($\Delta P(e) = 18$ W, $\Delta P(rg) = 13$ W). Then, considering only power, the optimal values of e and rg are their maxima values, but one should keep in mind that we can increase the stroke and power consequently - just as much as the machine volume allows it. Maximum value of rg is

$$rg = \min \left(d_2 + \frac{d_4}{2} - e, d_1 + \frac{d_3}{2} - e \right). \quad (53)$$

It is noticed that strokes of displacer and working piston are equal and vary in the same way while changing e and rg. Shown in Figure 7c,d,e,f are the respective effects

of the rhombic drive parameters d_1 , d_2 , d_3 , and d_4 on indicated power. These figures indicate that increasing the value of any of these parameters has negative effect on power. The same reasoning as for e and rg could be done. Figure 8c,d,e,f gives the explanation of the former fact; any increase in d_1 , d_2 , d_3 , and d_4 values decreases strokes of both displacer and working piston, then dead volumes will increase, and total volume variation will be decreased, and consequently, the indicated power will be reduced. It is noticed that for the studied machine, for the same variation of d_1 and d_2 ($\Delta(d_1) \approx \Delta(d_2) \approx 0.0273$ m), the negative effect of d_1 is greater three times than that of d_2 ($\Delta P(d_1) = -18$ W, $\Delta P(d_2) = -6$ W). This is due to the fact that the working piston is related to bar d_1 ; thus, its effect is much more important than that of bar d_2 , which is related to the displacer which has no other role but moving gas from one side to another, whereas the working piston has an effect on total volume variation, directly related to indicated work and power. Also, for bars d_3 and d_4 , similar remarks and conclusion could be done. For the same variation, the effect of d_3 is greater two times than that of d_4 for the

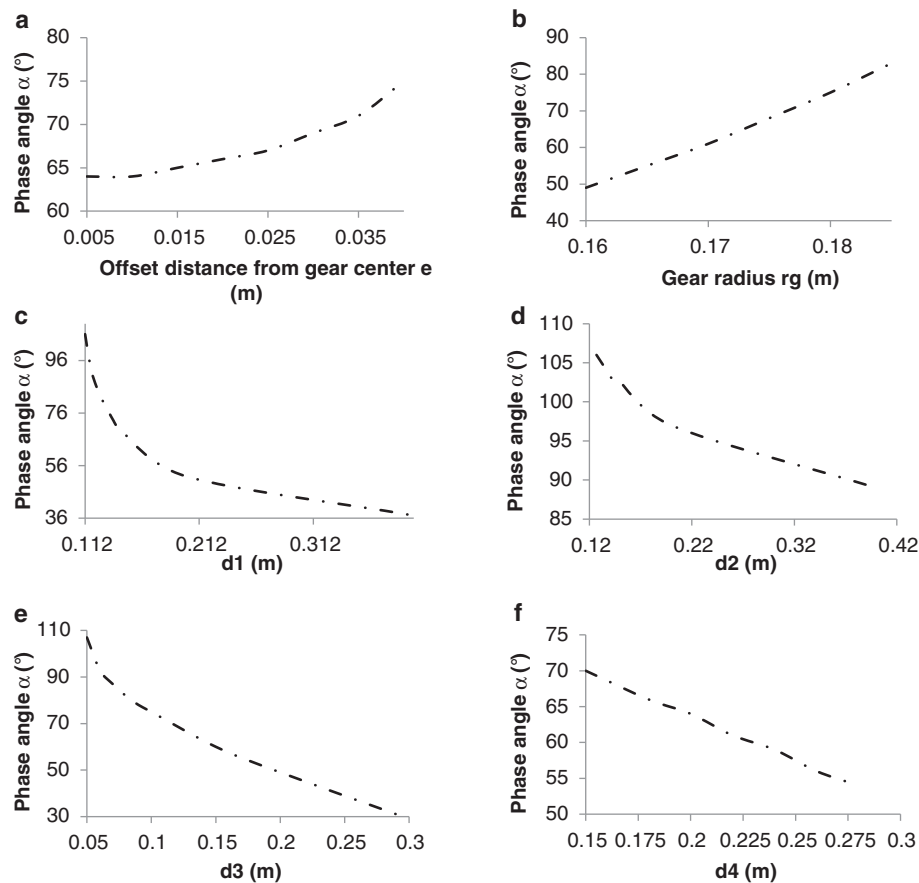


Figure 9 Effect of rhombic drive parameters on phase angle.

studied prototype. Therefore, for increasing the machine power, minima values of these parameters should be adopted:

$$d_1 = e + rg - \frac{d_3}{2} \quad (54)$$

$$d_2 = e + rg - \frac{d_4}{2} \quad (55)$$

$$d_3 = 2(e + rg - d_1) \quad (56)$$

$$d_4 = 2(e + rg - d_2). \quad (57)$$

Figure 8 indicates that parameters d_1 and d_3 have an effect only on working piston stroke, but no influence on displacer stroke, while parameters d_2 and d_4 have an influence only on displacer stroke and no effect on working piston stroke. For parameters e and rg , they have simultaneous equal effect on both displacer and working piston strokes. This gives a directive about adjusting the stroke of one of the pistons (for example, for reducing dead volume) without changing the stroke of the other. Many investigations about optimal phase angle for beta-type Stirling engines show that the optimal value is 90° [16].

Figure 9 shows the influence of different parameters of the rhombic drive mechanism on phase angle. As noticed for power, the increase in values of the offset distance from the gear center (e) or gear radius (rg) increases the phase angle, whereas increasing the rest of the rhombic drive parameter values decreases the phase angle. It is noticed that for the same variation of e and rg , the effect of gear radius rg is greater four times than the effect of e . Also, the effect of parameter d_1 is greater three times than that of d_2 on the phase angle of the studied prototype, and the effect of d_3 is greater two times than that of d_4 . This fact is due to the tightness of working space versus displacer-related space, which makes the working piston parameters having more considerable influence on the phase angle than that of the displacer-related parameters. As a conclusion, for obtaining a special phase angle, one should pay a special attention to parameters rg , d_1 , and d_3 .

Conclusions

Theoretical analysis has been performed to study the rhombic drive mechanism parameter effect on a low-temperature solar Stirling engine. The variation of the machine spaces was investigated, and a thermodynamic analysis and a calculation of power based on Schmidt theory were conducted. A parametric study was then performed to evaluate the effect of each rhombic drive

mechanism parameter on some machine characteristics, like power, piston and displacer strokes, and phase angle. In accordance with the obtained results of this study, offset distance from the gear center and gear radius have to be maximized in order to maximize the indicated power, while other rhombic drive parameters should be reduced for the same reason. Their minima values were given relatively to each other. Directives about realizing the optimal phase angle of beta-type Stirling engines were also given. As a conclusion, based only on rhombic drive optimization, one can increase the studied prototype power about 50%.

Competing interests

The authors declare that they have no competing interests.

Authors' contributions

HEH performed the rhombic drive and the thermodynamic studies. JK designed and manufactured the studied prototype. NB supervised the research, performed the Schmidt calculation, and helped JK on constructing the studied machine. MEH supervised the research. All authors read and approved the final manuscript.

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